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**ANALYSIS OF NON-STATIONARY PROCESSES
OF HEAT AND MASS EXCHANGE
IN THE RECIPROCATING COMPRESSOR SUCTION SYSTEM
FOR NATURAL GAS COMPRESSION**

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Abstract

Mobile gas filling stations are being created to increase the effectiveness of the use of compressed natural gas for automobile transport. The equipment of such stations consists of the storage accumulators (1-10 batteries of gas balloons), reciprocating compressor, which is used for better discharging of automobile reservoirs, as well as long pipelines systems.

The present research work shows the influence of suction pressure and temperature, as well as the geometric parameters of suction system on the effectiveness of the reciprocating natural gas compressor. This investigation was fulfilled with using mathematical simulation of compressor work.

Introduction

A considerable difference between the designed and real values of effectiveness parameters was revealed as a result of the experimental station exploitation, and first of all: compressor productivity and capacity. Calculations carried out in the course of designing works for reciprocating compressor model /1,2/ operating in typical for Russia winter conditions showed a considerable increase of compressor capacity of the first stage. As a result, calculated distribution of pressure for the stages, supercharging gas temperature and compressor indicator power differ from the real ones. The analysis showed a number of peculiarities, which were not taken during the simulation and which are typical for such compressor installations: long pipelines (diameters 10-20mm), joint (JC) and buffer (BC) cameras for decrease gas pulsations, the work in summer time as well as in winter, variable suction pressure (2-15 MPa), reality of compressing natural gas.

Depending on the compressor (CU) connection with a certain accumulator (AKi) (Fig.1) the length (Li) of suction pipeline will be different. It is known that the capacity depends on pressure pulsation, caused by transient gas flow in the suction system. The frequency and amplitude of the oscillation of gas pressure and temperature depend on its characteristics and geometrical parameters of the system. In its turn, the pulsation may influence compressor power and capacity both to increase and to reduce the values.

In the condition of low temperature (-30°C and less) while compressor works a considerable difference appears in the suction system between gas temperature in the inlet section of the pipeline and the temperature in suction camera in front of the valves. When the pressure of suction gas is from 2 to 20 MPa, the influence of suction gas heating on compressor capacity will be increased by the characteristics of compressing gas. For example: for the temperature of -23°C (suction conditions) and the pressure of 14 bar the compressibility coefficient (Z) is .65, and for the temperature of $+27^{\circ}\text{C}$ (approximately corresponds to gas parameters in the section in front of the suction valves of working compressor) is .82.

Model

Mathematical model is a dynamic model for gas flowing in pipeline system /3/ and in the stage of reciprocating compressor. The main correlations for the processes in compressor stage are shown in /1/. Mathematical model of gas flow in pipelines, which is based on the equations of nonstationary one-dimensional gas dynamics in partial derivatives. As the main variables, as well as for the equations of compressor mathematical model, we can choose gas density (ρ), specific initial energy (u), pressure (p), temperature (T) and gas velocity (average for section) (W). The system of

equations in five unknowns (ρ , u , p , T , W) consists of the equation of mass conservation, the equation of momentum conservation, the equation of energy conservation, the equation of gas state and the equation which connects thermodynamic parameters. The last equations are defined in compliance with the method shown in /4/, and depend on the type of applied gas.

Special attention must be drawn to the boundary conditions. In acoustic tasks gas parameters for the open end of the pipeline are usually equal to gas parameters outside the pipeline, and gas velocity has its maximum value. Such characteristic of boundary conditions does not seem to be correct, because in this case the change of temperature and pressure, caused by kinetic energy changes of flowing gas and by heat exchange, is excluded. The boundary conditions are more logical if an interconnecting "half-infinite" zone near the open end of the pipeline is neglected. The assumption of processes quasistationary state gives the opportunity to connect gas parameters in pipeline with gas parameters in a certain distance from the end without a detailed consideration of non-stationary spacial gas flowing in this zone. We shall assume that gas velocity in a distance is equal to zero, i.e. stationary gas (Fig.2). The boundary conditions in this case in the integral form can be presented as the system of nonlinear equations in four unknowns $p(0)$, $T(0)$, $u(0)$, $r(0)$ and are as follows:

$$\begin{aligned} x=0 \\ p(0,t) &= p_0(t) - Dp - W^2(0,t)/2 ; \quad u(0,t) = i_0(t) - W^2(0,t)/2 - p(0,t)/r(0,t); \\ T(0,t) &= f(r(0,t), u(0,t)), \quad p(0,t) = Z(r(0,t), T(0,t))r(0,t)RT(0,t) . \end{aligned}$$

$$\begin{aligned} x=L \\ p(L,t) &= p_L(t) - Dp - W^2(L,t)/2 ; \quad u(L,t) = i_L(t) - W^2(L,t)/2 - p(L,t)/r(L,t); \\ T(L,t) &= f(r(L,t), u(L,t)), \quad p(L,t) = Z(r(L,t), T(L,t))r(L,t)RT(L,t) , \end{aligned}$$

where $p(0)$ and $p(L)$ - pressure in the adjacent parts, connected by the pipeline, R - gas constant, Dp - pressure losses in intake or outlet end of pipe. Gas velocities $W(0)$ and $W(L)$ are found by the use of extrapolational formula for calculated values of velocities in the inner points of the pipeline.

Mass flow of gas in any section of the pipeline is defined by the formula:

$$m(x,t) = r(x,t) \cdot F_{\text{pipe}} \cdot W(x,t).$$

In consideration of the processes of heat exchange it is assumed to use quasistationary representation of processes for the known law of distribution of temperature along the pipeline and of temperature of camera walls. Coefficients of heat exchange, as well as resistance coefficients are according to the formulas of the established laminar and turbulent gas flow. For the universality of the calculated program the equations of the real gas state in the form of polynomial relations are added to the general model of the systems. To take into account thermodynamic characteristics of real gas it is recommended to use specific inner energy and gas density as the main variables. The applied numerical method /3/ in combination with the accepted scheme of construction of the general mathematical model allows not to use in calculations such gas parameters as heat capacity, volume and temperature exponents of isentropic extension, sound velocity. There is no necessity to include the procedures of differential equations of gas state to work out the calculation programs.

The compressor for natural gas compression /1/ was used as an object for the research. The following regime parameters will be chosen: suction pressure (p_s) - 3, 10 MPa, suction gas temperature (T_s) - 260K (-13°C), 310K (+27°C), discharging pressure - 25 MPa. During the research work the length of the pipeline was changed up to 6m, that corresponds to the real construction of gas filling station.

Results

Let us consider some of the calculation results according to the proposed method.
GAS PARAMETERS As first of all we are interested in the influence of the pipeline on compressor work, we shall analyze the change of gas parameters in the section in front of the suction valve of the compressor first stage.

Fig.3,4 show time diagrams of the change of gas pressure ($P_{s.c.}$) and temperature ($T_{s.c.}$) in the suction camera in front of the valve. The character and amplitude of parameters oscillation are different for different values of pipeline length. For better understanding of the reasons Fig.5 shows time diagrams of the change of gas velocity in the intake and outlet sections of suction pipeline. These diagrams show very well that for the length of 6m there is the process with the first fundamental frequency of gas pulsation in the pipeline (the first resonance), and for the length of 2m - with the second fundamental frequency (the second resonance). While opening the valve in the outlet section of the pipeline there is gas rarefaction. Gas starts to move and flows into the suction camera. The time of impulse advancing from the outlet section to the inlet section depends on the length of the pipeline. Fig.5 shows it by the location of velocity diagrams for inlet and outlet sections. Thus, there is a phase lag for long pipelines, which is connected with the final velocity (sound velocity) of impulse distribution. The longer the pipeline, the bigger the phase lag. Gas velocity reaches the big value during the resonance. Besides that the gas flow may change its direction. Such oscillating process is followed by the increase of energy losses for gas pushing. For long pipelines the losses to suction reaches 25 and more percentage of compression capacity. For short pipelines ($L=1m$) nonstationary effects connected with gas lag are practically absent. The curve of gas velocity change repeats the dependence the instantaneous velocity of a reciprocating. The values of gas velocity at the suction and outlet are different as the gas temperature and density are different in these sections. The overfall of pressure for a short pipeline with account of hydraulic losses may be defined according to the formulas applied in [2].

The changes of gas pressure in time in the suction camera cause a considerable change of temperature. And for long pipelines gas temperature increases as a result of gas heating, that leads to the decrease of gas density and volume capacity.

GAS HEATING Fig.6 shows the influence of the pipeline length and regime parameters on the average temperature of gas (T_{av}) in the suction camera. It is clearly seen that gas is heated flowing through the suction system. The influence of gas heating during the suction process is increased for natural gas, as there is a considerable difference of characteristics from the characteristics of ideal gas. After the change of temperature of intaking gas caused by heating there is a change of compressibility coefficient (Fig.7). For natural gas these changes reach big values both as for the average values during the suction process and for the extreme ones. On Fig.7 the range of changes of compression coefficient is enlarged and is more than 10 % with the increase of pipeline length. It is natural that the real gas density and real productivity of compressor will be changed. Such influence is particularly seen for lower suction temperatures of gas and for high pressure. For the same suction system the compressibility coefficient changes from .75 to 1. in accordance with gas regime parameters.

COMPRESSOR EFFECTIVENESS Let us estimate the influence of suction pipeline on the compressor work effectiveness. Fig.8,9 shows the dependence of volume capacity and indicator power of the compressor first stage from the pipeline length and regime parameters.

As it was expected, the influence of intaking gas temperature on volume capacity is not essential. The value and character of relations for different temperature of suction gas (Fig.8) for pressure $p=3MPa$ practically coincide.

The increase of volume productivity during the increase of suction pressure is explained by the fact that though the intermediate pressure is growing, but the pressure ratio increasing in the compressor first stage, which eliminate the influence of idle space.

It is necessary to pay attention to the influence of the pipeline length on compressor productivity.

The enclosed diagrams show that the decrease of productivity may be more than 20 % for long pipelines. This phenomenon can be explained by considering pressure changes in suction camera (Fig.3). It is known from the theory of reciprocating compressors that a considerable pressure decrease in the moment of valve closing is typical for the first resonance, and therefore the decrease of productivity. For the second resonance the pressure exceeds the nominal suction pressure in the moment of valve closing, which leads to the increase of capacity. The given results confirm this fact.

VALVE The work of the valve seems to be interesting in connection with the pipeline length. It was fulfilled analysis of the diagrams of the movement of the poppet of suction valve for different length of pipelines. If the length $L=6\text{m}$ the suction valve is closing with a considerable delay. The impact velocity in the moment of closing increases more than 2 fold and exceeds the accepted values. This result shows once again that the dynamic processes in the suction system influence the work of the valve.

JOINT CAMERAS Fig.11,12 show the results of simulation of processes when cameras (JC1 or JC1+JC2) joint to the pipe on certain distance (L_1, L_2) from compressor. It was considered three variants (Fig.1):

1 (pipe) - $L=2\text{m}$; 2 (pipe +JC1) - $L=2\text{m}, L_1=1\text{m}$; 3 (pipe +JC1+JC2) - $L=2\text{m}, L_1=1\text{m}, L_2=1.75\text{m}$. The main aim of such performance of suction system was decrease of gas pulsations. It is clearly seen that for third variants the amplitude of gas pulsations decreases. Such as the results show that suggested method allows to fulfill the correct solution of this problem.

Conclusions

1. While estimating the reciprocating natural gas compressor effectiveness it is necessary to take into account the main geometrical characteristics of the suction systems : pipeline length, diameter and volume joint or buffer cameras.
2. The use of long pipelines causes a considerable gas heating in front of the suction valve, eliminating compressor capacity. For high suction pressure and low temperature the heating leads to a considerable change of compressibility coefficient, which also influences the value of compressor productivity.
3. Real construction of the suction system may have a big influence on valves operation.
4. The analysis of the operation of the described compressor showed that the type of gas flow, including resonance regimes, practically does not depend on the value of the average suction pressure.

Literature

1. B.S.Chrustalev, V.B.Zdalinsky, V.A.Bulanov MATHEMATICAL MODEL OF RECIPROCATING COMPRESSOR WITH ONE OR SEVERAL STAGES FOR THE REAL GASES //Proceedings.1996 International Compressor Engineering Conference at Purdue, USA. Vol.1. P.211-216
2. B.S.Chrustalev, V.B.Zdalinski Die mathematische Modellierung der Arbeitsprozesse im vielstufigen Kolbenverdichter// Kolloquium Entwicklung und Betrieb von Kolbenverdichern TUD, Dresden,1996
3. Perevozchikov M.M., Chrustalyov B.S. Theoretical and Experimental Researches of Unsteady Gas Flow in the Pipe of the Reciprocating Compressor//Proceedings. 1994 International Compressor Engineering Conference at Purdue, USA. Vol.2. P.515-520.
4. B.S.Khrustalev Extension of Capabilities of the SIMCOP program for Quality Evaluation of the Working Processes in Reciprocating Compressors in the Design Stage// Chemical and Petroleum Engineering, USA, New York, July, 1996

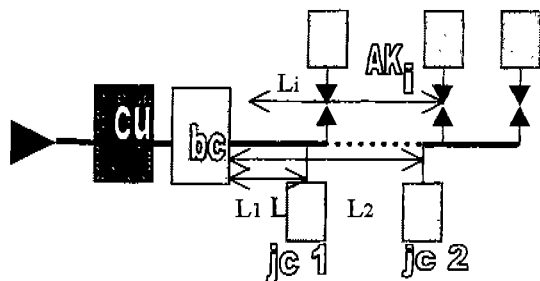


Fig.1 General scheme of suction system of gas filling installation.

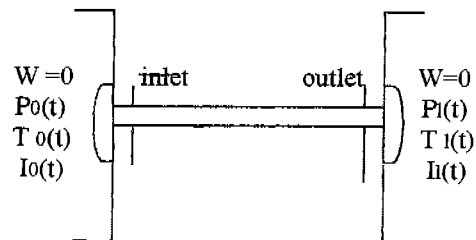


Fig.2 Boundary conditions for pipe

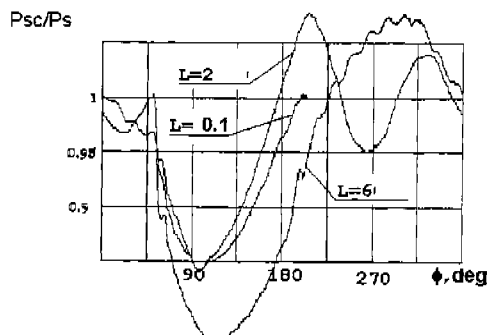


Fig.3 Pressure in suction camera of compressor

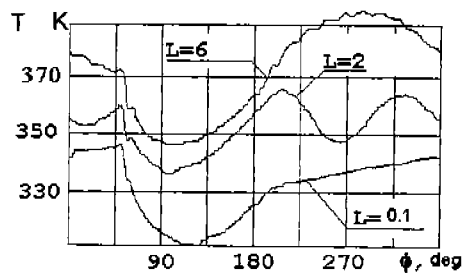


Fig.4 Temperature in suction camera of compressor

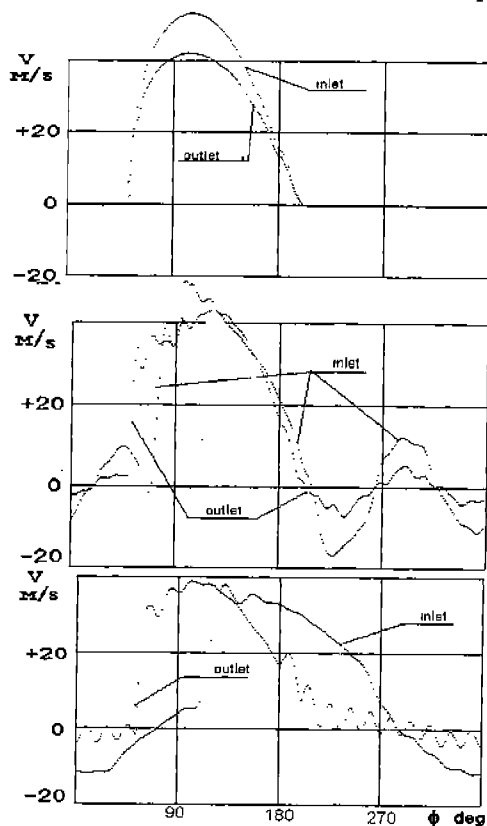


Fig.5 Gas velocity in the inlet and outlet section of the pipe.

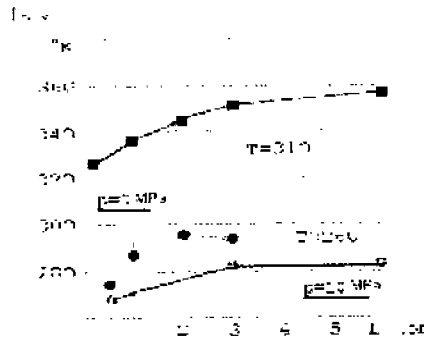


Fig. 6 Average gas temperature suction camera

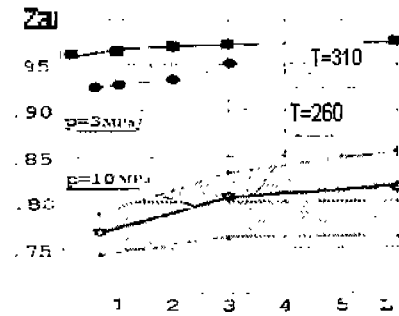


Fig. 7 Variation of compressibility coefficient of suction camera

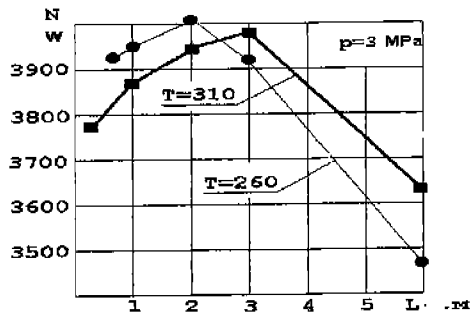


Fig. 8 Indicator power of the first compressor

stage

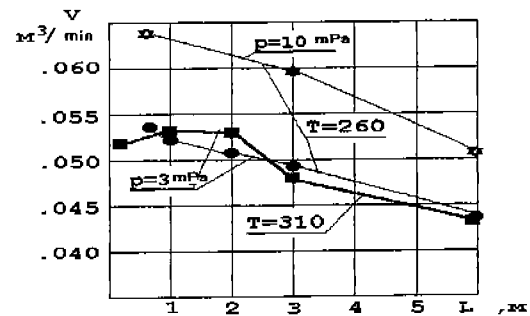


Fig. 9 Volume capacity of compressor.

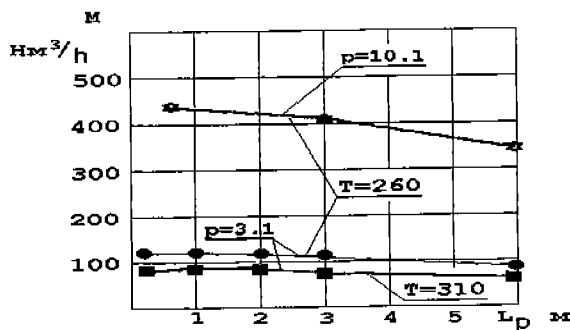


Fig. 10 Diagram of the suction valve of the first stage.

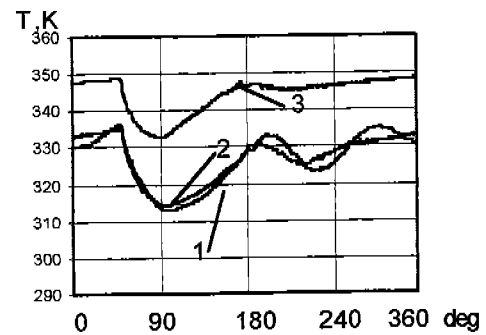


Fig. 11 Temperature in suction camera (1- pipe, 2- pipe+ jc1, 3- pipe+ jc2)

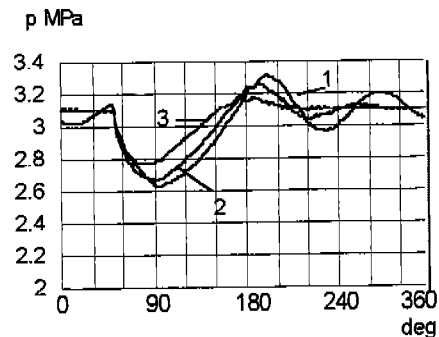


Fig. 12 Pressure in suction camera of compressor (1- pipe, 2- pipe+ jc1, 3- pipe+ jc2)